Multi-stage heat-pipe heat exchanger for improving energy efficiency of the HVAC system in a hospital operating room[1]

Ragil Sukarno[1], Nandy Putra[1,*], Imansyah Ibnu Hakim[1], Fadhil Fuad Rachman[1] and Teuku Meurah Indra Mahlia[2]

[1]Applied Heat Transfer Research Group, Department of Mechanical Engineering, Universitas Indonesia, Kampus UI, Depok, 16424, Indonesia; [2]School of Information, Systems and Modelling, Faculty of Engineering and Information Technology, University of Technology Sydney, NSW 2007, Australia

Abstract

The demands of specific requirements related to thermal comforts, such as temperature, relative humidity, inside air exchange and other factors required in a hospital operating rooms, have necessitated the development of energy-efficient heating, ventilation and air conditioning (HVAC) systems and efficient heat-recovery system using a heat-pipe heat exchanger (HPHE). The experiment was conducted by using HPHEs having three, six and nine rows, with four heat pipes in each row, arranged in a staggered configuration with a variation of fresh-air inlet temperature and velocity. The theoretical analysis was conducted using the ε-NTU method for predicting the effectiveness, outlet temperature of the evaporator side and energy recovery of the HPHE. The experimental results indicated that increasing the air-inlet temperature in the evaporator section and the number of rows increased the HPHE effectiveness but increasing the air-inlet velocity reduced the effectiveness. The highest effectiveness of 62.6% was obtained at an air-inlet temperature of 45°C with an air-inlet velocity of 2 m/s and a 9-row HPHE. The energy recovery of the HPHE increased with the number of rows, air-inlet temperature and air velocity in the evaporator section. The ε-NTU method can be used as a comparison method in the analysis of heat-recovery systems that apply HPHE air conditioning systems. Heat pipes that utilize cold-air exhaust from a room in an HVAC system can enhance efficiency and reduce emissions.

Keywords: heat-pipe heat exchanger; effectiveness; ε-NTU; energy recovery

*Corresponding author: nandyputra@eng.ui.ac.id

Received 12 September 2019; revised 29 May 2020; editorial decision 18 June 2020; accepted 18 June 2020

1. INTRODUCTION

With the increasing price of fossil fuels and their environmental problems, renewable energy and energy efficiency have become major aspects of the energy strategies of several countries. Many countries are attempting to increase the percentage share use of renewable energy, e.g. in the transportation sector [26], in the residential sector [2] and for electricity generation [12]. The main problem with renewable energy is its unstable supply; thus, renewable energy systems are always coupled with energy-storage equipment (usually batteries). Owing to the limitations of batteries, scientists are investigating alternative materials for energy storage [4, 16, 22]. Energy-efficiency programs in all sectors are important strategies for many countries [9, 11]. Intensive effort to reduce the energy growth on a global scale continues; however, dependency on fossil fuels as the main energy resource remains dominant. The increasing demand for facilities or new buildings that require maintenance of comfort levels will increase. Moreover, to provide comfortable rooms for occupants, buildings are usually equipped with heating, ventilation and air conditioning

1 This paper has been presented at the 18th International Conference on Sustainable Energy Technologies (SET) held in Kuala Lumpur, Malaysia, 20–22 August 2019.

International Journal of Low-Carbon Technologies 2020, 00, 1–9

doi:10.1093/ijlct/ctaa048

(HVAC) systems [3]. In commercial buildings, HVAC systems account for a significant share of the total energy consumption [19] and have been determined to be 40%–60% of the total [20, 21]. Rooms with special demands such as hospital operating rooms require specifications related to the temperature, relative humidity, exchange of inside air and other factors necessitate a more efficient HVAC system with low energy consumption.

According to ASHRAE, an operating room requires an inside air temperature that ranges from 20 to 24°C, a relative humidity that ranges from 30 to 60% and 15 to 20 air changes per hour [5, 6, 10, 28]. Further, the HVAC is often operated continuously. Therefore, heat recovery from the waste heat of HVAC systems represents an effort to use energy efficiently [8]. The use of a waste heat-recovery system with a heat pipe represents an excellent method for saving energy and reducing global-warming effects [24, 27, 29]. In recent years, heat-pipe technology has been increasingly used in the building service industry to enhance energy savings and the thermal performance of heat exchangers in commercial HVAC systems [14].

In HVAC systems, heat recovery can be enhanced by installing a heat-pipe heat exchanger (HPHE) on the air inlet and outlet of the ducting system through which air passes before reaching the cooling-coil device as a precooling device. Before air enters the cooling-coil device, the fresh-air temperature is reduced by recycling the cooled air from a room by using HPHEs. Thus, the energy consumption for cooling can be reduced, as well as the time needed to reach the dew-point temperature [1, 13, 15, 17, 21, 29].

Studies on the effects of different operating conditions and design parameters on thermal performance and energy saving, including both experimental and numerical analyses related to the development of the HPHE, have been performed by various researchers.

For example, Putra *et al.* [21] investigated the use of an HPHE as a precooling device in an HVAC system. The HPHE consisted of several pipes arranged in staggered configurations with various numbers of rows, including 2, 4 and 6, with a total of 39 pipes in 6 rows, and with water serving as the working fluid. The fresh-air inlet temperature was varied from 30 to 45°C under 1, 1.5 and 2 m/s flows of fresh air. The largest heat recovery achieved was 1404.29 kJ/h, with effectiveness of 0.15. The results indicated that increasing the number of rows of the tubes increased the effectiveness of the HPHE, and energy savings could be achieved [21]. Muhammaddiyah *et al.* [17] added multi-wavy fins to the evaporator and condenser sections of the HPHE that were applied in the HVAC system. An HPHE consisting of 42 pipes was arranged in a staggered fashion with three rows. The results indicated that the highest effectiveness of 54.4% was obtained at an air-inlet velocity and temperature of 1 m/s and 45°C, respectively [17]. The use of theoretical analysis based on the NTU-effectiveness (ε-NTU) method for predicting the effectiveness of a thermosyphon HPHE has been reported [18, 31], and the effectiveness of the thermosyphon HPHE was compared between experimental and theoretical results.

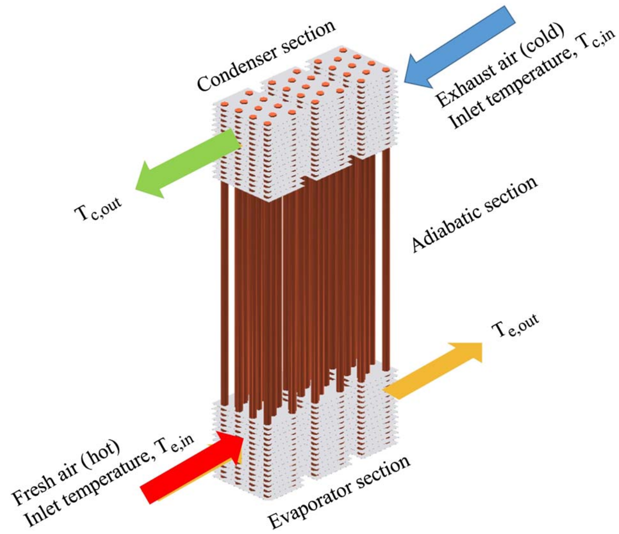

Figure 1. *HPHE design.*

The literature review indicates that the use of an HPHE reduces the energy consumption in HVAC systems, but new developments are needed to enhance the HPHE performance. Moreover, with the rapid growth of new hospitals, the number of operating rooms will increase. Hence, intensive efforts to save energy by recovering the waste heat from HVAC systems are important. The main objective of this study was to investigate the possible energy savings and effectiveness of an HPHE for heat recovery in the HVAC system used in hospital operating rooms. A performance analysis employing the ε-NTU method was conducted to predict the theoretical effectiveness of the HPHE. The results can be used to provide guidance for continuing to the next steps of fabrication and experimental testing.

2. METHODOLOGY

2.1. Test facility design

This study was performed using an experimental test model comprising a simulation room with mini-chiller and ducting systems. The ducting system consisted of one inlet and two outlet channels on the two sides of the simulator room. The HPHE consisted of several heat-pipe rows that were arranged in a staggered configuration. The number of rows was varied between three, six and nine, with four heat pipes in each row.

The heat-pipe tubes were made of copper, and the inner surface of the tubes contained a wick structure of sintered copper. The tubes were filled with water as the working fluid at a filling ratio of 50%. The outer diameter of the heat pipe was 10 mm, and the evaporator was 160 mm long. The length of the adiabatic section was 360 mm, and the length of the condenser section was 190 mm. To enhance the heat-transfer area, the HPHE was equipped with a continuous wavy fin made of aluminum with a thickness of 0.105 mm and a fin spacing of 2 mm, as shown in Figure 1.

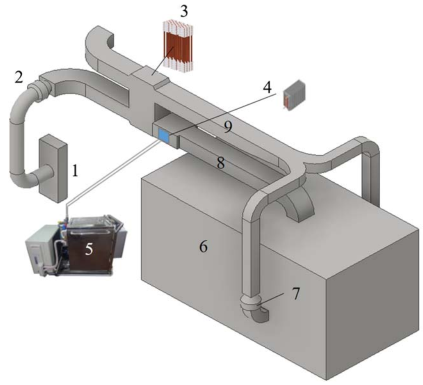

Figure 2. *Experimental test model.*

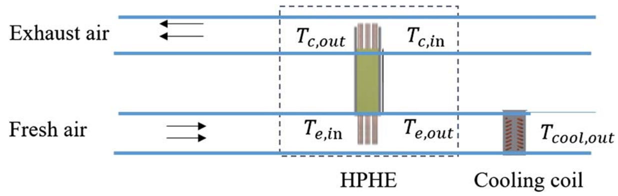

Figure 3. *Installation of the HPHE in the ducting.* [13, 21].

The experimental test model was equipped with system control and a measurement device, as shown in Figure 2. The heat input involved a 4000-W heater equipped with a proportional–integral–derivative temperature controller that was placed before the axial fan inlet to customize the fresh-air inlet temperature in the evaporator section. The mini-chiller device consisted of a circulating thermostatic bath, a cooling coil and a pump with a flow meter, which was used to deliver chilled water through the cooling coil that was mounted in the ducts of the air-handling unit.

2.2. Effectiveness

For describing the thermal performance of the HPHE, sensible effectiveness is a highly relevant parameter. Effectiveness is defined as the ratio of the actual heat transfer to the maximum heat transfer in a heat exchanger [21], which can be determined using Equation 1 for the test results obtained.

$$\varepsilon = \frac{Q_{act}}{Q_{max}} \quad (a) \qquad \text{or} \qquad \varepsilon = \frac{T_{e,in} - T_{e,out}}{T_{e,in} - T_{c,in}} \quad (b) \qquad (1)$$

2.3. Energy recovery

In HVAC systems, the evaporator side of the HPHE is installed on the inlet side of the ducting as a precooling device, while the condenser section is installed on the exhaust side of the air ducting system.

In HVAC systems without heat pipes, the fresh air is typically overcooled from $T_{e,\text{in}}$ to $T_{\text{cool, out}}$ until it reaches the dew-point temperature and is subsequently reheated for the dehumidification process [31]. The cooling load is determined using Equation 2. By adding the HPHE as a precooling device, the cooling load of the cooling coil of the mini-chiller can be distributed, as indicated in Equation 3 [31].

$$q_{\text{load}} = \dot{m}\, c_p \left(T_{e,\text{in}} - T_{\text{coil,out}}\right) \qquad (2)$$

$$q_{\text{load}} = q_{\text{recovery}} + q_{\text{cooling coil}} \qquad (3)$$

Thus, the cooling load of the cooling coil ($q_{\text{cooling coil}}$) can be expressed as follows:

$$q_{\text{cooling coil}} = \dot{m}\, c_p \left(T_{e,\text{out}} - T_{\text{coil,out}}\right) \qquad (4)$$

The energy recovery that can be achieved by applying the HPHE as a precooling device is given in Equation 5.

$$q_{\text{recovery}} = \dot{m}\, c_p \left(T_{e,\text{in}} - T_{e,\text{out}}\right) \qquad (5)$$

2.4. ε-NTU effectiveness

The ε-NTU effectiveness approach has been used to investigate the performance of a thermosyphon HPHE [14, 18, 31]. In an analysis using the ε-NTU approach, a thermosyphon HPHE is considered as two separate heat exchangers; i.e. the evaporator and condenser sections are associated with the working fluid in the heat pipe [18, 31]. By determining the fresh-air inlet temperature in the evaporator, air inlet in the condenser, surface temperature of the heat pipes, and HPHE geometry, ε-NTU analysis can be used to predict the performance of the HPHE and can be very helpful in the design stage of the HPHE. The vapor inside the heat pipe is almost at a constant temperature; thus, the vapor capacity rate C_{v} is equal to infinity, corresponding to a heat-capacity ratio, $C_e/C_{\text{v}} = C_c/C_{\text{v}}$, which is equal to zero. Assuming that $C_e/C_{\text{v}} = C_c/C_{\text{v}} = 0$, the ε-NTU of the HPHE at a single row of the evaporator and condenser can be expressed as in Equation 6 [14, 18, 31].

$$\varepsilon_{e,1} = 1 - e^{(-\text{NTU}_e)} \quad (a) \quad \text{and} \quad \varepsilon_{c,1} = 1 - e^{(-\text{NTU}c)} \quad (b) \quad (6)$$

The number of transfer units (NTU) in the evaporator and condenser sections is determined using Equation 7.

$$\text{NTU}_e = \frac{U_e A_e}{C_e} \quad (a) \quad \text{and} \quad \text{NTU}_c = \frac{U_c A_c}{C_c} \quad (b) \qquad (7)$$

For the NTU calculation, the areas of an HPHE with a number of rows of heat pipes (denoted as A_e and A_c) are based on the total heat-transfer area in a row [14]. The HPHE developed in this study is equipped with a continuous wavy fin; thus, fins are considered in determining the heat-transfer area. For the evaporator and condenser sections, the heat capacity C is calculated using Equation 8.

$$C_e = \dot{m}_e\, c_{p,e} \quad (a) \quad \text{and} \quad C_c = \dot{m}_c\, c_{p,c} \quad (b) \qquad (8)$$

The overall heat-transfer coefficient for the evaporator and condenser sections is calculated using the following equation [14].

$$\frac{1}{U_eA_e} = \frac{1}{\eta_{o,e}\ h_e\ A_{hp}} + R_{hp,e} \quad \text{(a)} \quad \text{and}$$
$$\frac{1}{U_cA_c} = \frac{1}{\eta_{o,c}\ h_c\ A_{hp}} + R_{hp,c} \quad \text{(b)} \tag{9}$$

The overall fin surface efficiency η_{so} η_o can be calculated using the following equation [7, 30].

$$\eta_{so} = 1 - \frac{A_f}{A_o}\ \left(1 - \eta_f\right) \tag{10}$$

Single-fin efficiency [30].

$$\eta_f = \frac{\tanh\left(mr\varphi\right)}{mr\varphi}, \tag{11}$$

where $m = \sqrt{\frac{2h}{kt}}$

For an HPHE with n rows, the ε-NTU effectiveness is given in Equation 12 [18, 31].

$$\text{In evaporator :} \quad \varepsilon_{en} = \frac{\left[\frac{1-\frac{C_e}{C_v}\varepsilon_{e,1}}{1-\varepsilon_{e,1}}\right]^n - 1}{\left[\frac{1-\frac{C_e}{C_v}\varepsilon_{e,1}}{1-\varepsilon_{e,1}}\right]^n - \frac{C_e}{C_v}}$$
$$\text{In condenser section :} \quad \varepsilon_{en} = \frac{\left[\frac{1-\frac{C_c}{C_v}\varepsilon_{c,1}}{1-\varepsilon_{c,1}}\right]^n - 1}{\left[\frac{1-\frac{C_c}{C_v}\varepsilon_{c,1}}{1-\varepsilon_{c,1}}\right]^n - \frac{C_c}{C_v}} \tag{12}$$

Because $C_e/C_v = C_c/C_v = 0$, Equation 12 can be simplified as follows [18, 31]:

$$\varepsilon - \text{NTU for } n \text{ rows} \tag{13}$$
$$\text{In evaporator section :} \quad \varepsilon_{en} = 1 - \left(1 - \varepsilon_{e,1}\right) \quad \text{(a)}$$
$$\text{Incondenser section :} \quad \varepsilon_{cn} = 1 - \left(1 - \varepsilon_{c,1}\right) \quad \text{(b)}$$

The overall effectiveness of the HPHE based on ε-NTU can be defined as follows [18, 31]:

$$\text{If } C_e > C_c: \quad \varepsilon_0 = \left[\frac{1}{\varepsilon_{c,n}} + \frac{\frac{C_c}{C_e}}{\varepsilon_{e,n}}\right]^{-1} \quad \text{(a)}$$

and

$$\text{If } C_c > C_e: \quad \varepsilon_0 = \left[\frac{1}{\varepsilon_{e,n}} + \frac{\frac{C_e}{C_c}}{\varepsilon_{c,n}}\right]^{-1} \quad \text{(b)} \tag{14}$$

In this study, the mass flow rate of air at the inlet of the evaporator section is 0.056, 0.075, or 0.094 kg/s lower than that in the condenser section. Thus, the heat capacity is $C_c > C_e$. Accordingly, the HPHE effectiveness with three, six and nine rows can be determined. After the effectiveness is obtained, the outlet air temperature in the evaporator section $T_{e,\text{out}}$ can be predicted as follows:

$$T_{e,\text{out}} = T_{e,\text{in}} - \varepsilon_o\ \left(T_{e,\text{in}} - T_{c,\text{in}}\right) \tag{15}$$

Determining the overall heat-transfer coefficient U requires the convection coefficients for the evaporator and condenser sections, i.e. h_e and h_c, respectively. The convection coefficients for the evaporator and condenser sections are obtained by considering the external cross flow in a bank of tubes. The thermal resistance of the heat pipe R is obtained from the measurements of 0.0041–0.0007°C/W.

The bank of tubes of the HPHE applied in the HVAC system is shown in Figure 4.

To determine the convection coefficients of the HPHE in the bank of tubes, the Zukauskas correlation can be used, as follows [7]:

$$\text{NU}_D = C_1\ C_2\ \text{Re}_{D,\max m}\ \text{Pr}^{0.36}\left(\frac{\text{Pr}}{\text{Pr}_s}\right)^{1/4} \tag{16}$$

The constant C_1 is the dimension that is associated with the configuration and the Reynolds number in the bank of tubes of the HPHE. C_2 is the correction constant for the heat exchanger with ≤ 20 rows of tubes. In this HPHE, the tube diameter D is 10 mm and is measured as the transverse pitch between tube centers. S_T is 25 mm, and the longitudinal pitch S_L is 21.65 mm. Thus, $S_T/S_L < 2$ and the constants $C_1 = 0.36$ and $m = 0.6$ are obtained. In contrast, the constant C_2 varies according to the number of rows of the HPHE ($C_2 = 0.84$ for three rows, $C_2 = 0.94$ for six rows and $C_2 = 0.96$ for nine rows).

The Reynolds number $R_{e,D}$ used in Equation 16 is based on the maximum velocity that occurs in the evaporator and condenser sections. A staggered configuration on an external cross flow is shown in Figure 4, with $S_D = (S_T + d)/2$. The maximum velocity is determined using the following equation.

$$V_{\max} = \frac{S_T}{S_T - d} V_{e,in} \tag{17}$$

The convection coefficient is obtained from the Nusselt number through the following expression [7, 23].

$$h = \frac{Nu.k}{d} \tag{18}$$

According to the temperature difference obtained from the experiment, the thermal resistance of the heat pipe can be obtained by dividing the temperature difference by the heat-transfer rate, as

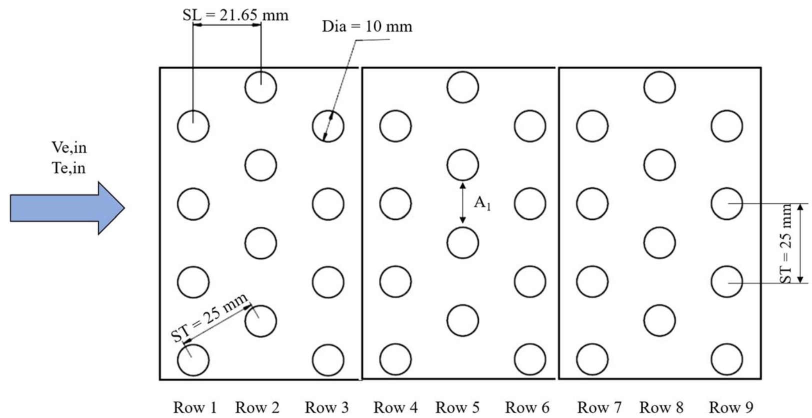

Figure 4. *Schematic of tube arrangements in a bank with nine rows in a staggered configuration.*

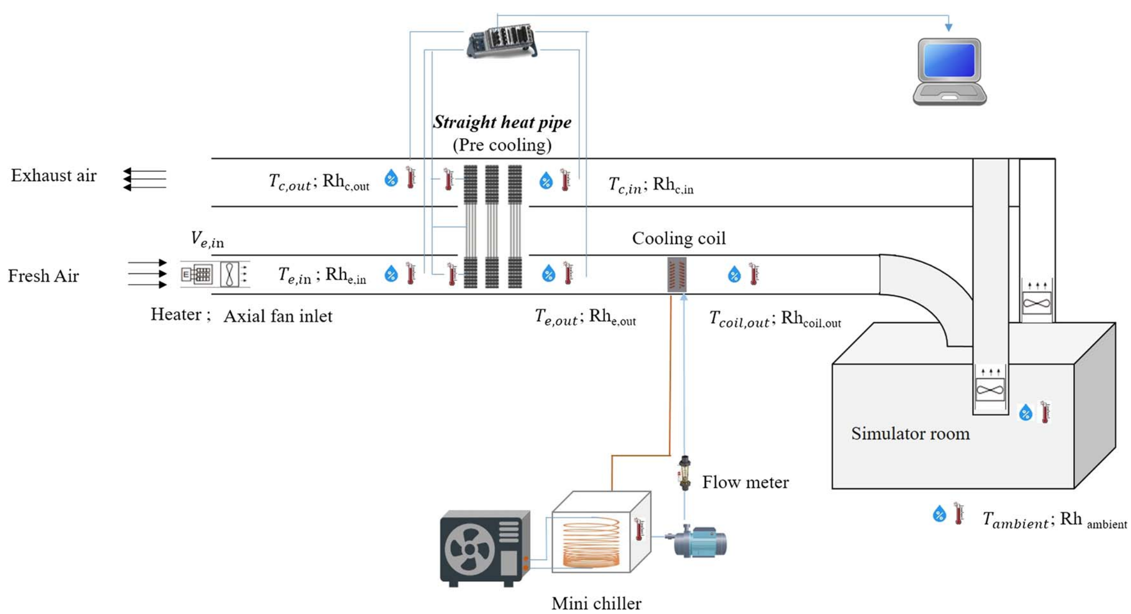

Figure 5. *Experimental setup.*

indicated in Equation 18 [23, 25].

$$R = \frac{T_{hp,e} - T_{hp,c}}{\dot{Q}} \tag{19}$$

2.5. Experimental setup

The performance of the HPHE is influenced by several parameters, including design and operation parameters.

The air temperatures at the inlet and outlet of the evaporator and condenser section were measured with type-K thermocouples which were connected to module NI 9214 with accuracy ±0.01°C, and the relative humidity at the same locations was measured using a humidity sensor (Autonic THD), which was connected in series to a data-acquisition device (NI cDAQ-9174). The measurement points on the HPHE shown in the experimental schematic are presented in Figure 5. The mass flow rate of the chilled water in the cooling coil in a constant guard was 4 L/min. The air-inlet velocities of the evaporator and condenser sections were measured using hotwire anemometer Lutron AM-4204 sensors with accuracy ± (5% + 0.1 m/s) at the center of the duct.

3. RESULTS AND DISCUSSIONS

3.1. HPHE performance

A series of tests were conducted to investigate the performance of the HPHE arranged in a staggered fashion with three, six and nine rows. The fresh-air inlet temperature in the evaporator section was varied between 30, 35, 40 and 45°C, and the velocity of the inlet air in the evaporator section was varied between 1.5, 2.0 and 2.5 m/s. The reading for each variation was obtained at intervals of 30 min after steady-state conditions were achieved, and the steady-state conditions were achieved after the prototype

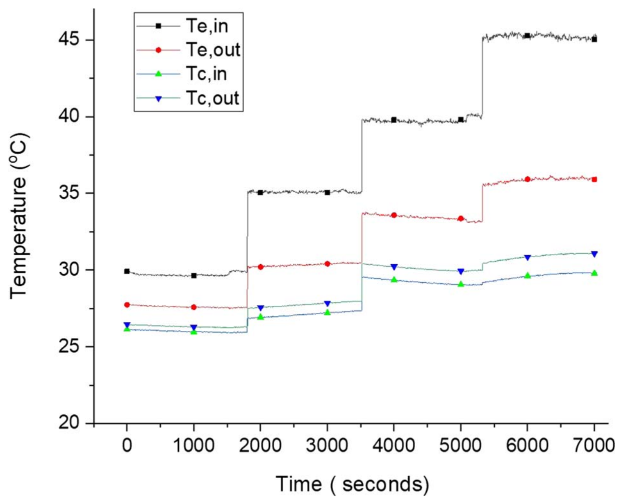

Figure 6. *Temperature profile for an inlet air velocity of 1.5 m/s and nine rows.*

was run for 10 min. Figure 6 displays the air-temperature profile on the inlet and outlet for the evaporator and condenser sections separately with an incoming air velocity of 1.5 m/s in the nine-row HPHE.

Figure 6 shows that fresh-air temperature decreases from the inlet side of the evaporator ($T_{e,in}$) to the outlet side of the evaporator HPHE ($T_{e,out}$). When the fresh-air inlet temperature was 30°C, the fresh-air temperature decreases by 2.1°C. With increasing fresh-air inlet air temperature, the fresh-air temperature difference between the inlet and outlet sides of the HPHE evaporator will increase, and when the fresh-air inlet temperature was 45°C, the outlet of fresh-air temperature decreases by 9.4°C. It shows that the utilization of the HPHE to the HVAC system was successfully used as the precooling media. The HPHE significantly absorbed the heat of the water inlet in the evaporator section, reducing the temperature before the air entered the cooling-coil devices. Heat release occurred through the cold air that passed through the condenser section of the HPHE. The air outlet temperature on the side of the condenser section was higher than with that in the inlet section. Figure 6 shows that with an increase in the fresh-air temperature in the evaporator section, the amount of heat absorbed by the heat pipe increased. This finding is indicated by the increased temperature difference between the inlet and outlet of the evaporator section.

The test results for investigating the effects of the fresh-air inlet temperature, the air velocity and the number of rows on the effectiveness are shown in Figure 7. The results indicate that at the same velocity, increasing the fresh-air inlet temperature and the number of rows increased the effectiveness. With the same number of rows, the effectiveness decreased with an increase in the air velocity. The highest effectiveness (62.6%) was obtained with an air-inlet temperature of 45°C, an air-inlet velocity of 2 m/s and nine rows of the HPHE. The lowest effectiveness (43.1%) was obtained with an air-inlet temperature of 30°C, an air-inlet velocity of 2 m/s and three rows of the HPHE.

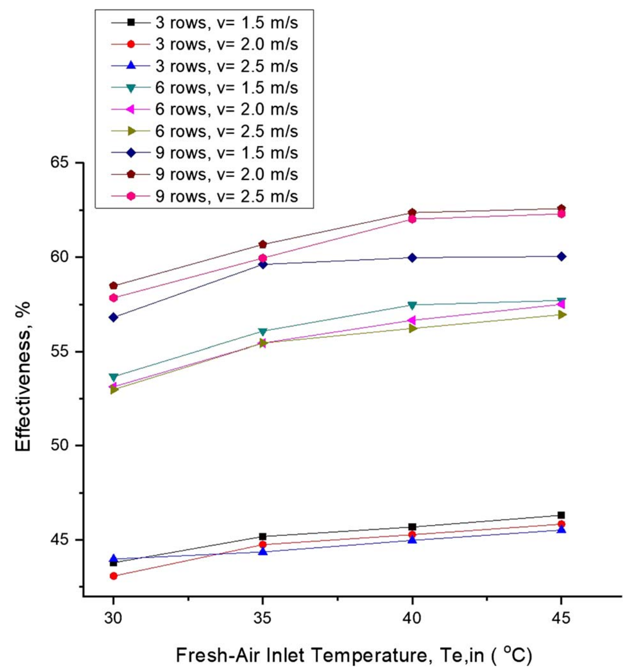

Figure 7. *HPHE effectiveness profile with respect to the number of rows, air-inlet temperature and velocity.*

In conventional HVAC systems, the fresh air is cooled by a cooling coil device from ($T_{e,in}$) to the supply air temperature ($T_{cool,out}$). When the HPHE was applied, the fresh-air temperature can be reduced before it was cooled by a cooling-coil device, so that the cooling energy by the chiller decreased. With the application of the nine-row HPHE in the HVAC system and in the condition which a fresh-air inlet temperature of 45°C and air velocity of 1.5 m/s, the fresh-air temperature was decreased by 9.4°C, and energy recovery obtained was 533 W. Energy recovery indicates that heat is absorbed by the HPHE before it is cooled by the cooling-coil device. The largest temperature difference was 9.8°C, which was obtained with an air-inlet temperature of 45°C, an air velocity of 2.5 m/s and nine rows of the HPHE. Under these conditions, the maximum energy recovery was 931.6 W. The smallest temperature difference in the evaporator section was 1.4°C, which was obtained with an air-inlet temperature of 30°C, an air velocity of 1.5 m/s and three rows of the HPHE. Under these conditions, 82.1 W of energy was recovered. The effects of the fresh-air inlet temperature, the air velocity and the number of rows on the energy recovery are shown in Figure 8.

Figures 6–8 show that increasing the fresh-air-inlet temperature increased the effectiveness and energy recovery. The ability of the heat pipe to absorb heat was affected by the boiling point of the working fluid in the heat pipe. As the temperature of the inlet fresh air entering the evaporator section increased, the rate of evaporation of the working fluid in the heat pipe increased, and the heat absorption increased.

The effectiveness and energy recovery decreased with increasing air velocity. This indicates that reducing the air velocity improved the heat absorption of the HPHE. Additionally, these

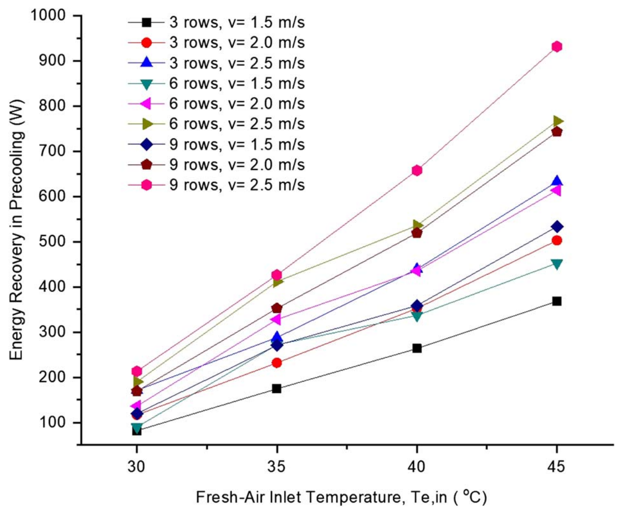

Figure 8. *Energy-recovery profile with respect to the number of rows, fresh-air-inlet temperature and velocity.*

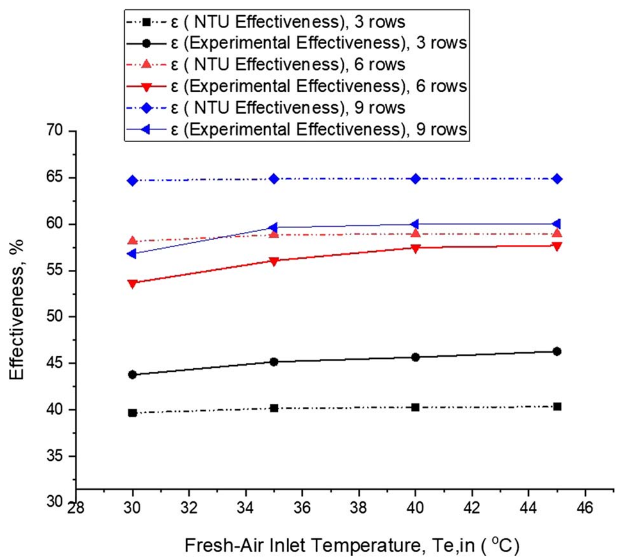

Figure 9. *Comparison of the experimental effectiveness and ε-NTU analysis results for different fresh-air inlet temperatures and row numbers at a velocity of 1.5 m/s.*

results indicate that the precooling process of the fresh-air temperature in the evaporator occurred, and the temperature was significantly reduced before entering the cooling-coil device. Thus, the amount of energy needed to cool the air was reduced, and the dew point could be achieved more rapidly. The air returned from the room that was still cold was recycled and entered the condenser section to absorb heat, causing the working fluid to condense back into a liquid phase.

3.2. Performance comparison of experimental and ε-NTU effectiveness

Figure 9 shows a comparison between the experimental effectiveness and the predicted overall effectiveness obtained from the

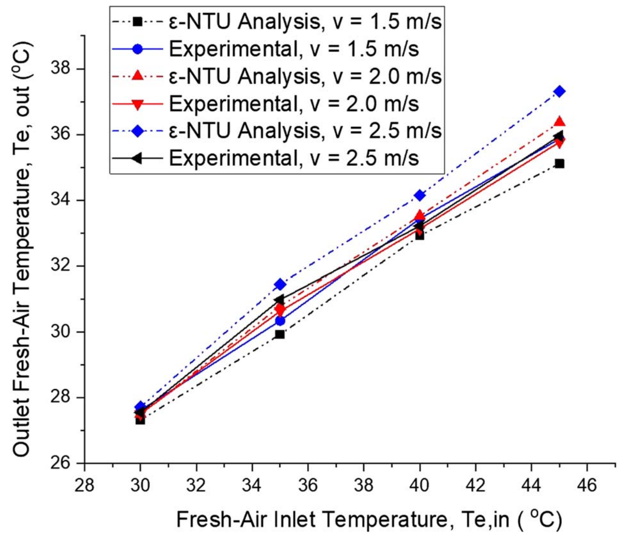

Figure 10. *Comparison of the temperature profile in the evaporator section for nine rows with different fresh-air inlet temperatures and velocities.*

ε-NTU analysis for different fresh-air inlet temperatures, velocities and row numbers.

After the effectiveness was obtained (and when the fresh-air inlet temperatures in the evaporator and condenser were known), the inlet air temperature in the evaporator section could be specified, as shown in Figure 10. For both of these methods, the theoretical predictions were similar to the experimental results. The effect of the heat load of the evaporator section on the thermal resistance of heat pipes is shown in Figure 11. The heat load and repeated energy recovery of the precooling process obtained by applying the HPHE were determined using Equation 5. With a heat load of 82.1–931.6 W, the thermal resistance decreased to 0.0041–0.0007°C/W. Increasing the heat load in the evaporator section reduced the thermal resistance of the heat pipe. Greater heat absorption was obtained when the air-inlet temperature increased. Thus, increasing the air-inlet temperature on the evaporator side reduced the thermal resistance, increasing the effectiveness of the HPHE.

4. CONCLUSIONS

According to the results, the following conclusions are drawn.

- An increase in the air-inlet temperature on the evaporator side and an increase in the number of rows increased the HPHE effectiveness, but an increase in the inlet air velocity reduced the effectiveness.
- The highest effectiveness of 62.6% was obtained with an air-inlet temperature of 45°C, an air-inlet velocity of 2 m/s and nine rows of the HPHE. The lowest effectiveness of 43.1% was obtained with an air-inlet temperature of 30°C, an air-inlet velocity of 2 m/s and three rows of the HPHE.
- The energy recovery indicated that heat was absorbed by the HPHE before the air was cooled by the cooling-coil device. The

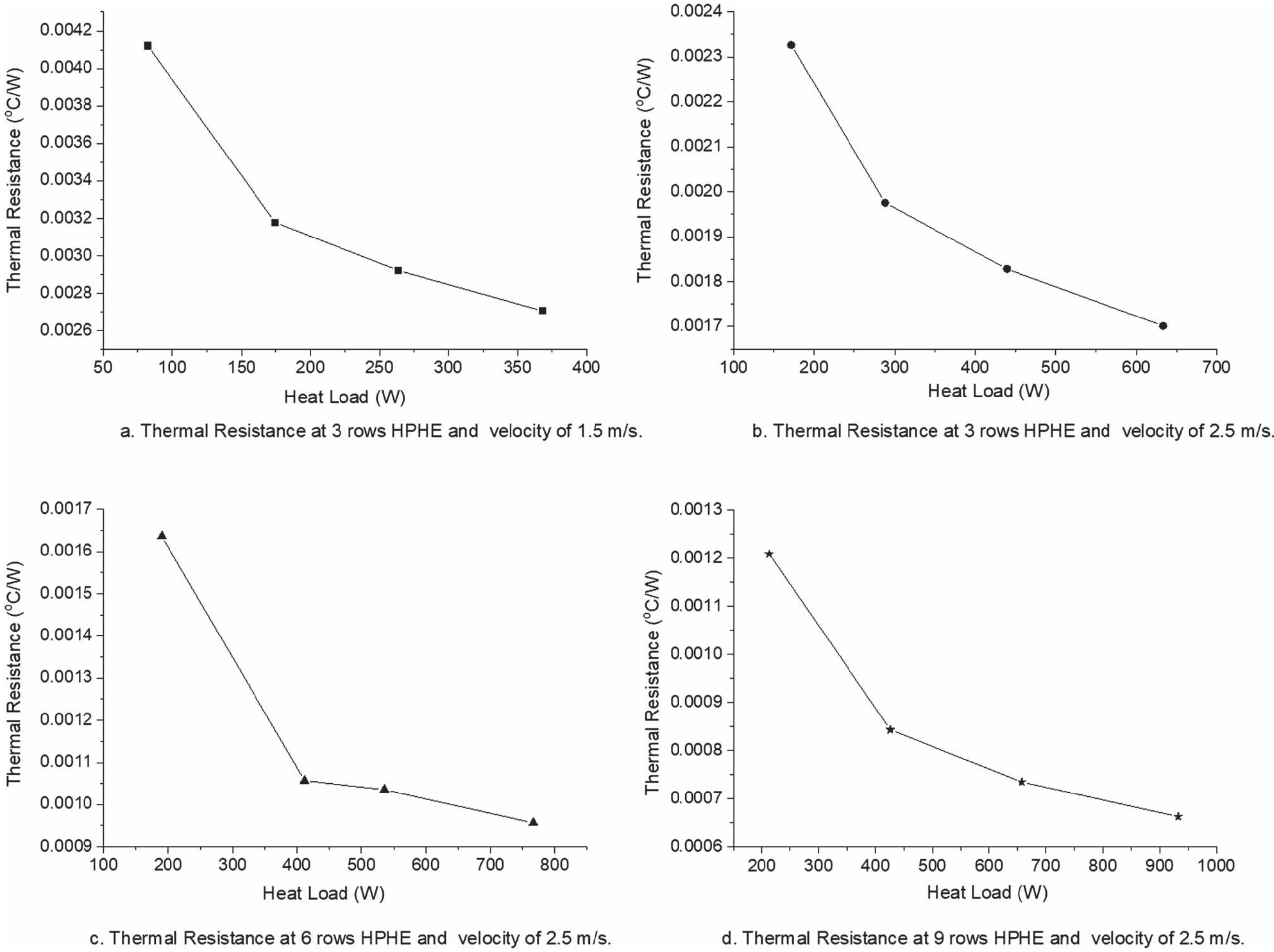

Figure 11. *Effect of the heat load to thermal resistance of the HPHE.*

largest temperature difference was 9.8°C, which was obtained at an air-inlet temperature of 45°C, an air velocity of 2.5 m/s and nine rows of the HPHE. Under these conditions, the maximum energy recovery was 931.6 W. The smallest temperature difference in the evaporator section was 1.4°C, which was obtained with an air-inlet temperature of 30°C, an air velocity of 1.5 m/s and nine rows of the HPHE. Under these conditions, 82.1 W of energy was recovered.

- The ε-NTU method can be applied to HPHE systems for predicting the effectiveness of the HPHE. A theoretical analysis using the ε-NTU method provided results similar to those obtained via the experiment, and this method can be used as a comparison method for analysis of heat-recovery systems using an HPHE. Thus, the method can be used as an initial reference to predict the performance of an HPHE design before the design and manufacturing steps.
- Increasing the heat load in the evaporator section reduced the thermal resistance of the heat pipe. Increased heat absorption was obtained when the air-inlet temperature was increased. These findings indicate that increasing the air-inlet temperature on the evaporator side reduced the thermal resistance, hence increasing the HPHE effectiveness.

ACKNOWLEDGEMENTS

The authors would like to thank the financial support from Kemenristek Dikti through PPUPT Scheme 2019 with contract number NKB-1751/UN2.R3.1/HKP.05.00/2019.

REFERENCES

[1] Abd El-Baky MA, Mohamed MM. Heat pipe heat exchanger for heat recovery in air conditioning. *Appl Therm Eng* 2007;**27**:795–801.
[2] Aditya L, Mahlia TMI, Rismanchi B, NG, H. M., Hasan, M. H., Metselaar, H. S. C., Muraza, O. & Aditiya, H. B. A review on insulation materials for energy conservation in buildings. *Renew Sust Energ Rev* 2017;**73**: 1352–65.
[3] Ahmadzadehtalatapeh M. An air-conditioning system performance enhancement by using heat pipe based heat recovery technology. *Sci Iran* 2013;**20**:329–36.
[4] Amin M, Putra N, Kosasih EA *et al.* Thermal properties of beeswax/ graphene phase change material as energy storage for building applications. *Appl Therm Eng* 2017;**112**:273–80.
[5] ASHRAE. 2013. *HVAC Design Manual for Hospitals and Clinics*. Atlanta: ASHRAE Inc.
[6] Balaras CA, Dascalaki E, Gaglia A. HVAC and indoor thermal conditions in hospital operating rooms. *Energ Buildings* 2007;**39**:454–70.

[7] Bergman TL, Incropera FP, Dewitt DP, Lavine AS. 2011. *Fundamentals of Heat and Mass Transfer*. New Jersey: John Wiley & Sons, Inc.

[8] Cuce PM, Riffat S. A comprehensive review of heat recovery systems for building applications. *Renew Sust Energ Rev* 2015;**47**:665–82.

[9] Fournier ED, Federico F, Porse E, Pincetl S. Effects of building size growth on residential energy efficiency and conservation in California. *Appl Energ* 2019;**240**:446–52.

[10] Hakim, I. I., Putra, N., Marda, A. P., Alvaro, M. A. & Winarta, A. Experimental study on utilization of heat pipe heat exchanger for improving efficiency of clean room air system in hospitals. *E3S Web of Conferences*, 2018. vol. 67. Les Ulis: EDP Sciences, 02056.

[11] Han L, Han B, Shi X *et al.* Energy efficiency convergence across countries in the context of China's belt and road initiative. *Appl Energ* 2018;**213**:112–22.

[12] Ismail MS, Moghavvemi M, Mahlia TMI. Genetic algorithm based optimization on modeling and design of hybrid renewable energy systems. *Energ Convers Manag* 2014;**85**:120–30.

[13] Jadhav TS, Lele MM. Theoretical energy saving analysis of air conditioning system using heat pipe heat exchanger for Indian climatic zones. *Eng Sci Technol Int J* 2015;**18**:669–73.

[14] Jouhara H, Merchant H. Experimental investigation of a thermosyphon based heat exchanger used in energy efficient air handling units. *Energy* 2012;**39**:82–9.

[15] Kusumah A, Hakim I, Sukarno R *et al.* The application of U-shape heat pipe heat exchanger to reduce relative humidity for energy conservation in heating, ventilation, and air conditioning (HVAC) systems. *Int J Technol* 2019;**10**:1202.

[16] Mehrali M, Latibari ST, Mehrali M *et al.* Preparation and characterization of palmitic acid/graphene nanoplatelets composite with remarkable thermal conductivity as a novel shape-stabilized phase change material. *Appl Therm Eng* 2013;**61**:633–40.

[17] Muhammaddiyah S, Winarta A, Putra N. Experimental study of multi-fin heat pipe heat exchanger for energy efficiency in operating room air systems. *Int J Technol* 2018;**2**:422–9.

[18] Noie SH. Investigation of thermal performance of an air-to-air thermosyphon heat exchanger using ε-NTU method. *Appl Therm Eng* 2006; **26**:559–67.

[19] Papadopoulos S, Kontokosta CE, Vlachokostas A, Azar E. Rethinking HVAC temperature setpoints in commercial buildings: The potential for zero-cost energy savings and comfort improvement in different climates. *Build Environ* 2019;**155**:350–9.

[20] Pérez-Lombard L, Ortiz J, Pout C. A review on buildings energy consumption information. *Energ Buildings* 2008;**40**:394–8.

[21] Putra N, Anggoro T, Winarta A. Experimental study of heat pipe heat exchanger in hospital HVAC system for energy conservation. *Int J Adv Sci Eng Inf Technol* 2017;**7**:871.

[22] Putra N, Rawi S, Amin M *et al.* Preparation of beeswax/multi-walled carbon nanotubes as novel shape-stable nanocomposite phase-change material for thermal energy storage. *J Energ Stor* 2019;**21**:32–9.

[23] Ramos J, Chong A, Jouhara H. Experimental and numerical investigation of a cross flow air-to-water heat pipe-based heat exchanger used in waste heat recovery. *Int J Heat Mass Transf* 2016;**102**:1267–81.

[24] Reay D, MCglen R, Kew P. 2013. *Heat Pipes: Theory, Design and Applications*. Oxford: Butterworth-Heinemann.

[25] Shabgard H, Allen MJ, Sharifi N *et al.* Heat pipe heat exchangers and heat sinks: Opportunities, challenges, applications, analysis, and state of the art. *Int J Heat Mass Transf* 2015;**89**:138–58.

[26] Silitonga AS, Masjuki HH, Ong HC *et al.* Evaluation of the engine performance and exhaust emissions of biodiesel-bioethanol-diesel blends using kernel-based extreme learning machine. *Energy* 2018;**159**: 1075–87.

[27] Srimuang W, Amatachaya P. A review of the applications of heat pipe heat exchangers for heat recovery. *Renew Sust Energ Rev* 2012;**16**:4303–15.

[28] Teke A, Timur O. Assessing the energy efficiency improvement potentials of HVAC systems considering economic and environmental aspects at the hospitals. *Renew Sust Energ Rev* 2014;**33**:224–35.

[29] Vakiloroaya V, Samali B, Fakhar A, Pishghadam K. A review of different strategies for HVAC energy saving. *Energ Convers Manag* 2014;**77**:738–54.

[30] Wang CC, Fu WL, Chang, C. T. Heat transfer and friction characteristics of typical wavy fin-and-tube heat exchangers. *Exp Therm Fluid Sci* 1997;**14**:174–86.

[31] Yau Y, Ahmadzadehtalatapeh M. Heat pipe heat exchanger and its potential to energy recovery in the tropics. *Therm Sci* 2015;**19**:20–20.